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Flow Studies in a Centrifugal Compressor Stage

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ABSTRACT

The overall efficiency of the compressor is dependent on the design of both the impeller and diffuser. The vane diffuser reduces the operating range. However, by proper setting of the diffuser with reference to impeller, it is possible to achieve a good stage performance. This study is on the experimental investigation of the detailed flow behavior inside a centrifugal compressor stage for three different setting angles of the vane diffuser with reference to the fixed impeller blade outlet angle. It is seen that diffuser setting angles lower than the impeller outlet flow angle gives higher stage efficiency as well as flow range.

1. INTRODUCTION

Centrifugal compressors have applications in gas turbine engines, auxiliary power plants, remote pilot less vehicles, refrigeration plants, chemical and process industries, aircraft engine starters and turbo super chargers used in internal combustion engines. In a centrifugal compressor, work is imparted to the impeller to get higher stagnation pressure of the working fluid. A diffuser is employed at the downstream of the impeller for the conversion of kinetic energy of the flow coming out of the impeller into static pressure. The overall efficiency of the compressor is dependent on the design of both the impeller and diffuser. The vane diffuser reduces the operating range. However, by proper setting of the diffuser with reference to impeller, it is possible to achieve a good stage performance. The setting angle of the diffuser (matching) with reference to the impeller plays a crucial role on the stage performance.

W. Stein, M. Routenberg[1] carried out detailed measurements of the flow field inside a vane diffuser, a calculation method is verified and afterward employed to calculate the flow field inside a variable-geometry vane diffuser. Compared these results to measured lines of the compressor stage, and found that the computer performance characteristic of the diffuser fits the measurements. N. Arndt, A.J. Acosta, C.E. Brennen, T.K. Caughey[2] carried out experimental investigation of rotor-stator interaction in a centrifugal pump with several vane diffusers. The experiments were conducted for different flow coefficients and different radial gaps between the impeller blade trailing edge and the diffuser vane leading edge and found that the largest pressure fluctuations on the diffuser vanes and the impeller blades to be of the same order of magnitude as the total pressure rise across the pump. The largest pressure fluctuations on the diffuser vanes were observed to occur on the suction side of the vane near the vane leading edge, whereas on the impeller blades the largest fluctuations were observed to occur at the blade trialing edge.

This paper is on experimental investigations in a high pressure ratio centrifugal compressor stage consisting of impeller and vane diffuser with three different blade setting angles of 70, 65 and 60 degrees with reference to the radial direction were carried out by detailed steady flow measurements using conventional pressure and temperature sensors. The experiments were carried out at different speeds ranging from 15000 to 20000 rpm in a closed circuit centrifugal compressor test rig. Static pressure measurements were carried out on shroud wall from impeller inlet to diffuser exit to study the flow behavior in the compressor stage at different flow conditions. The experimental data were collected using an online data acquisition system and the parameters like mass flow rate, pressure ratio, efficiency and pressure recovery were estimated from the measured data to get the performance map. Static pressure measurements on the suction and pressure surface of the vaned diffuser and in channel from diffuser leading edge to diffuser exit at different radius were measured to study the variation of diffuser blade loading on the overall stage performance.

2. EXPERIMENTAL SETUP

2.1 Test Facility

The layout of the Closed Circuit Centrifugal Compressor Test Rig (CLOCTER) is shown in Fig. 1. An electromechanically coupled twin DC motor system rotates the compressor at a desired speed. Thyristor control with feedback for the DC motors ensures maintenance of the speed to an accuracy of 1%. The compressor and DC motor were connected together with a step up gear box (1:20). An electronic torque meter coupled in between the gear box and the compressor was used to measure compressor speed and input power. A gate valve provided in the closed circuit was used to vary the mass flow rate through the compressor. An orifice plate in the inlet duct was used for a mass flow measurement. A heat exchanger in the closed circuit was used to ensure steady inlet flow conditions. An external cooling tower system pumps cold water to this heat exchanger and to another heat exchanger, which cools the lubrication system. The orifice plate in the facility is used to monitor the flow rate, which can be varied by throttling the gate valve. This provides us to get wide range of operating points. Similarly a Thyristor control through feed back circuit is used for the variation of speed of the DC motor as this allows running the compressor to any desired speed.

Fig. 1: Closed Circuit Centrifugal Compressor Test Rig (CLOCTER)

2.2 Test Compressor

The compressor stage consisting of impeller and vane diffuser is shown in Fig. 2. The backswept impeller has 300 mm tip diameter having 19 blades and a vane diffuser having 17 blades. The diffuser inlet to impeller outlet diameter ratio is around 1.05, and the diffuser inlet to diffuser outlet radius ratios are 1.5 for D1, 1.511 for D2 and 1.555 for D3 configurations. The experiments were carried out by running the compressor at speeds ranging from 15000 to 20000 rpm in steps of 1000 rpm.

Fig. 2: Centrifugal Compressor Stage

2.3 Instrumentation

The time averaged parameters like total pressure, static pressure, total temperature, speed and power input to the compressor were measured using conventional probes and an on-line data acquisition system. The data acquisition system is connected to an industrial computer. Analog signals from the transducer and thermocouples were sequentially scanned and stored in the scanner. A 48 port scanivalve with a single pre-calibrated Statham transducer of 50 PSIA range with sensitivity 146.6 μv/v/psi was used for the measurement of pressure at various locations. The scanning rate of the scanivalve was controlled using a decoder. In the present case the system provides a scanning rate of 10 pressures per second, so that 48 pressures can be scanned in 4.8 seconds. The pressures were measured to an accuracy of 0.1% full scale reading.

3. RESULTS AND DISCUSSION

The performance characteristics of the impeller and the compressor stage were obtained by running the compressor in closed circuit with air as working medium. Different operating points were obtained for speeds ranging from 15000 to 20000 rpm in steps of 1000 rpm by varying the mass flow rate using a main and an interconnecting gate valve. The pressure ratio and mass flow rates were estimated from the measured time averaged parameters. The performance characteristics of the impeller and stage with three different configurations of the diffuser are shown in Figs. 3 and 4. It is observed from Fig. 3, that the operating range of the impeller varies with the diffuser blade setting angles. It is also observed, as the operating range increases the pressure ratio reduces at all speeds ranging from 15000 to 20000 rpm. With decrease in diffuser setting angle the diffuser throat area increases, this allows the compressor to swallow higher mass flow, hence the operating range of the stage increases at lower setting angle. At the same time the back pressure to the impeller reduces with lower blade loading. Hence the pressure ratio drops.

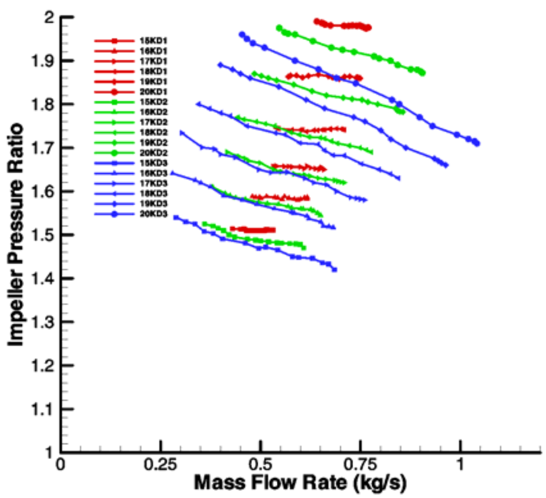

Fig. 3: Performance Characteristics of Impeller

Fig. 4 shows the stage characteristics for higher diffuser setting angle becomes steeper indicating larger pressure loss in the diffuser. At higher speeds the flow through the diffuser is chocked. The lowest mass flow rate at each speed indicates the stalled condition. It is observed that the stalled mass flow decreases as the diffuser setting angle is reduced. It is also observed that, it is possible to change the slope of the characteristics by varying the diffuser inlet angle to flow coming out of the impeller.

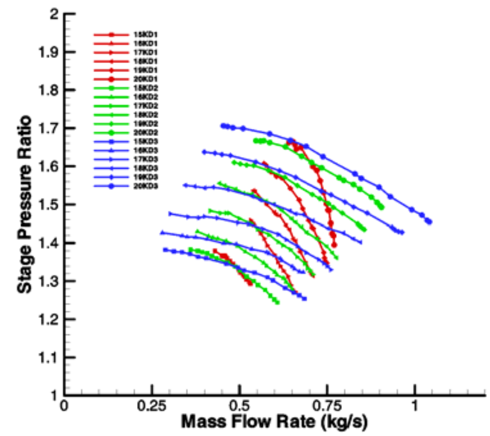

Fig. 4: Performance Characteristics of Stage

The pressure recovery in vane diffuser at D1, D2 and D3 configurations are plotted in Fig. 5 for different speeds ranging from 15000 to 20000 rpm. It is observed from this figure, that the diffuser with D3 configuration has the higher pressure recovery coefficient and the pressure recovery coefficient more or less remains same for wider operating range. Diffuser with higher setting angle the pressure recovery coefficient tends to zero or becomes negative. At high flow coefficients it indicates that the flow over the diffuser walls may be separated. It is observed that, with decrease in setting angle the performance of the diffuser improves. The pressure recovery coefficient becomes very sensitive with change in mass flow rate at higher setting angle of the diffuser. At lower setting angle and lower mass flow rates the pressure recovery coefficient at higher speeds are lower than that at lower speeds.

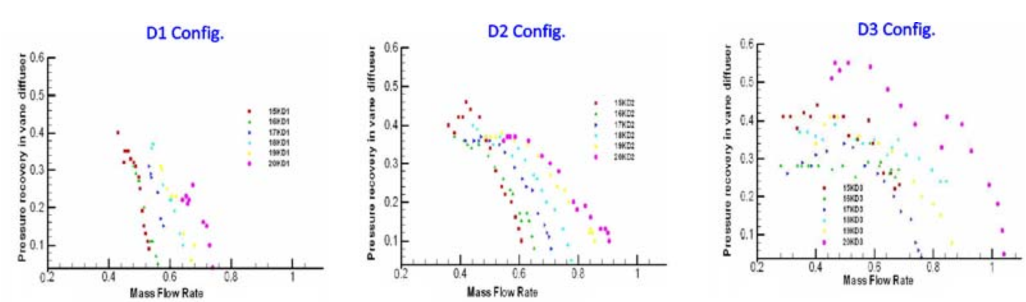

Fig. 5: Pressure Recoveries in Vane Diffusers at D1, D2 & D3 Configurations

The impeller and stage efficiencies for one of the configuration of the diffuser, namely D2 is shown in Fig. 6 for speeds ranging from 15000 to 20000 rpm. Similar characteristics were also obtained for the other configurations. The static pressure measurements in the diffuser channel are measured at different radius from diffuser leading edge to diffuser exit in configurations D1, D2 and D3 and the normalized static pressure contours with reference to inlet static pressure are plotted for four different mass flow rates at 20000 rpm and are shown in Fig. 7. It is observed from this figure, for a given setting angle the static pressure inside the diffuser channel increases with decrease in mass flow rate. The increase in static pressure at a given mass flow rate is higher for lower setting angle. Since the cone angle of the diffuser increases, this provides higher diffusion.

The variation of the static pressure across the diffuser wall is more uniform for higher setting angles of the diffuser indicating lower pressure gradient and less susceptible to flow separation. At lower setting angle the pressure gradient increases from W_2 to W_1 and flow has a tendency to separate from wall W_1 very near to exit.

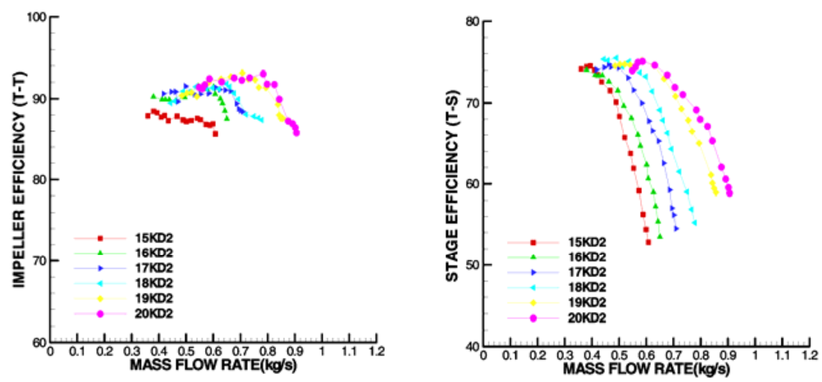

Fig. 6: Impeller and Stage Efficiencies at D2 Configuration

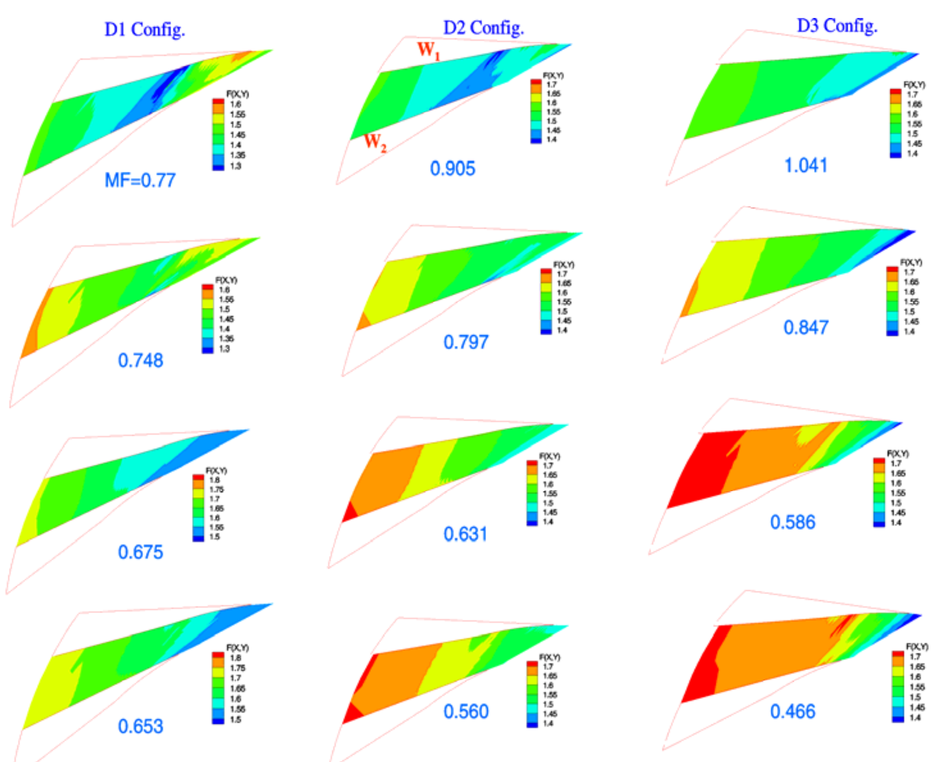

Fig. 7: Static Pressure Distribution in Diffuser Channel at 20000 rpm

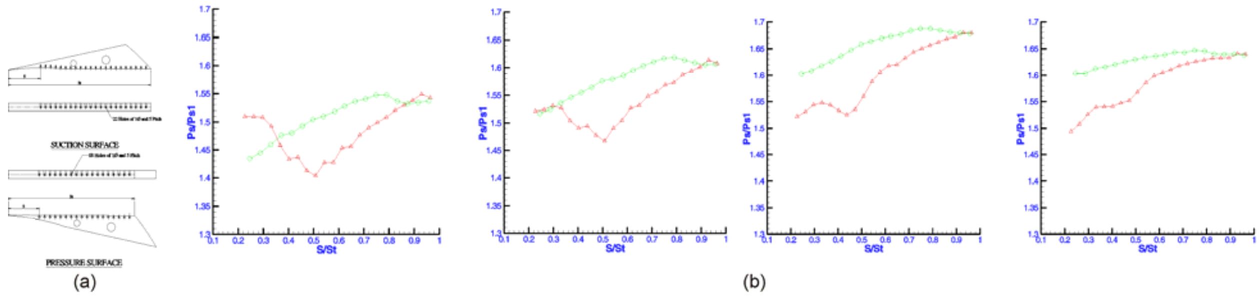

Fig. 8: Static Pressure Variation at Suction and Pressure Surface of Diffuser D2 at 20000 rpm

The vane diffuser provides static pressure recovery in a compressor stage. This recovery depends on the incidence to the diffuser. To study this, the static pressures were measured at suction and pressure surface wall of the vaned diffuser Fig. 8(a) in three configurations of the diffuser at different speeds and mass flow rates. This static pressure is normalized with corresponding inlet static pressure at different operating points. Variations of the static pressure along the blade length for four mass flow rates at 20000 rpm are plotted in Fig. 8(b). Suction surface pressures are indicated by the symbol "o" and the pressure surface pressures are indicated by the symbol "Δ". The static pressure variation on the diffuser blade is shown slightly away from the leading edge due to the difficulty in providing the pressure taps close to the leading edge. At high flow rates the incidence to the diffuser is positive and the flow might separate from the suction surface, due to this it is observed that there is a small cross over in the pressure distribution and as the flow rate reduces this cross over vanishes and the diffuser provides the required loading which gives rise to increase in static pressure. Throughout the flow regime the pressure surface values are higher than the suction surface, except at high flow rates close to the leading edge. It is also seen very close to the trailing edge both the pressur**es** becomes equal. Similar observations were made at other setting angles. The area within the pressure distribution curves indicate the diffuser blade loading. The estimated area under the pressure distribution curves indicates the area increases with decrease in setting angle indicating higher pressure rise.

4. CONCLUSIONS

Diffuser setting angle influences the impeller performance. The operating range of the impeller varies with the diffuser blade setting angles. As the operating range increases the pressure ratio reduces. With lower setting angle the impeller and stage efficiency decreases. The pressure recovery coefficient becomes very sensitive with change in mass flow rate at higher setting angle of the diffuser. At lower setting angle and lower mass flow rates the pressure recovery coefficient at higher speeds are lower than that at lower speeds. The variation of the static pressure across the diffuser wall is more uniform for higher setting angles of the diffuser indicating lower pressure gradient and less susceptible to flow separation. Diffuser setting angle lower than the impeller outlet flow angle, gives higher stage efficiency as well as flow range. At lower setting angle, the stage efficiency is higher over wider mass flow and the peak stage efficiency does not change with setting angle.

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